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Friction torque in a modified angular contact ball bearing operating at low axial loads

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Abstract. Friction torque in an angular contact ball bearing is dependent on the motion of the balls, lubricant viscosity and quantity, rotational speed and applied loads. To evaluate the friction torque in a ball bearing, the SKF methodology includes the geometrical parameters, lubricant viscosity and type of lubrication, operation speeds and applied loads (radial and axial). As is indicated in the SKF catalogue, the above-mentioned methodology can be used for applied loads equal to or larger than the recommended minimum load indicated. For very low applied loads and for dry or limit conditions, the SKF catalogue doesn’t indicate the adequate models to estimate the friction torque. Also, the presence of the cage is not directly included in the SKF methodology. Based on the spin down methodology, the authors performed experimental investigations on a 7205B modified angular contact ball bearing, by using only three balls with and without the cage. Experimental friction torques on the outer race has been obtained in dry conditions and the results were compared with the results obtained from the SKF catalogue. Important differences have been obtained. Also, it was evidenced that at very low loads the presence of the cage leads to an increase in the friction torque with more than one order of magnitude.

1.Introduction

In the evaluation of the friction torque for a rolling bearing, the friction generated by the rolling elements and cage is generally ignored for normal operating conditions (moderate speed, moderate loads, and presence of the lubricant). Thus, the SKF [1] methodology has been used to estimate with accuracy the friction torque in a rolling bearing as function of type, rotational speed, lubricant quantity and viscosity, lubrication method, applied loads and geometrical dimensions of the rolling bearings. Some of the rolling bearings can operate at various conditions, such as dry or limit conditions, very low loads, and low rotational speed and in these circumstances the evaluation of the friction torque is more complicated than the standard methodology. The research realized by Olaru et al. [2] on a 51100 thrust ball bearing evidenced that at very low loads, the friction between the cage and balls has an important contribution to the total friction torque in the thrust ball bearing. The authors used the spin–down methodology applied on a modified 51100 thrust ball bearing having only three balls, operating in dry conditions, with a very low axial load (1.26 N) and at a rotational speed between 60 and 210 rpm. To evidence the influence of the cage on the total friction torque, the authors realized two types of experiments in dry conditions: experiments with three balls without cage and experiments with three balls included in a steel cage. The presence of the cage leads to important increases of the friction torque (from $2*10^{-5}$ Nm without cage to $9*10^{-5}$ Nm in the presence of the cage). In the

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presence of lubricant, the influence of the cage on the friction torque in a modified 51100 thrust ball bearing, having only three balls, has been evidenced experimentally and demonstrated theoretically by Olaru *et al.* [3]. Therefore, it was evidenced that at a very low axial load (1.26 N), by using some drops of oil in the modified 51100 thrust ball bearing, the friction torque increases compared to dry conditions, as result of hydrodynamic forces developed both in the ball-race and in ball-cage contacts.

Balan et al. [4] conducted experimental studies on the influence of the cage on a 51205 standard thrust ball bearing, operating at a very low axial load (4.26 N) and between 100 and 400 rpm. The experiments evidenced that, for an oil viscosity of 0.35 Pas, the hydrodynamic friction between the balls and a standard steel cage leads to increases of the friction torque with (10-40)%.

Houpert [5, 6] developed a complex methodology to evaluate the friction torque in a radial and angular contact ball bearings based on the equilibrium of the forces and moments acting on the balls. In his methodology, Houpert includes both the friction force and friction moment between balls and cage. Unfortunately, the friction between the balls and cage is dependent on a lot of parameters (speed, geometry, lubricant) and it’s not so easy to be analytically determined.

Based on the previous methods used by Olaru et al. for the modified thrust ball bearings, an experimental methodology to determine the friction torque in 7206 B modified angular contact ball bearing has been developed, considering only the ball-races contacts. Also, the influence of the cage in relation to the friction torque in an angular contact ball bearing has been determined, for a very low axial load and in dry conditions. Supplementary, the authors simulated the total friction torque in 7206 B modified angular contact ball bearing by using the SKF methodology and compared the simulated results with the experimental values.

2.Experimental background

2.1. Modified 7205 B angular contact ball bearing

Figure 1 presents the modified 7205 B angular contact ball bearing used for the tests. In the first tests the angular contact ball bearing had only three balls, positioned at 120 degrees. The inner race of the ball bearing is fixed on the rotating table of the Tribometer CETR UMT2. On the outer race, a free rotating weight cylinder was fixed. The load applied by the cylinder and outer race imposed, on each ball-race contact, a normal force of Q=6.443N. The modified 7205B angular contact ball bearing has the following characteristics: inner diameter d=25 mm, outer diameter D=52 mm, ball diameter d_b=7.928 mm, inner race conformity f_i=0.515, outer race conformity f_e=0.522 and contact angle α=15 degrees.

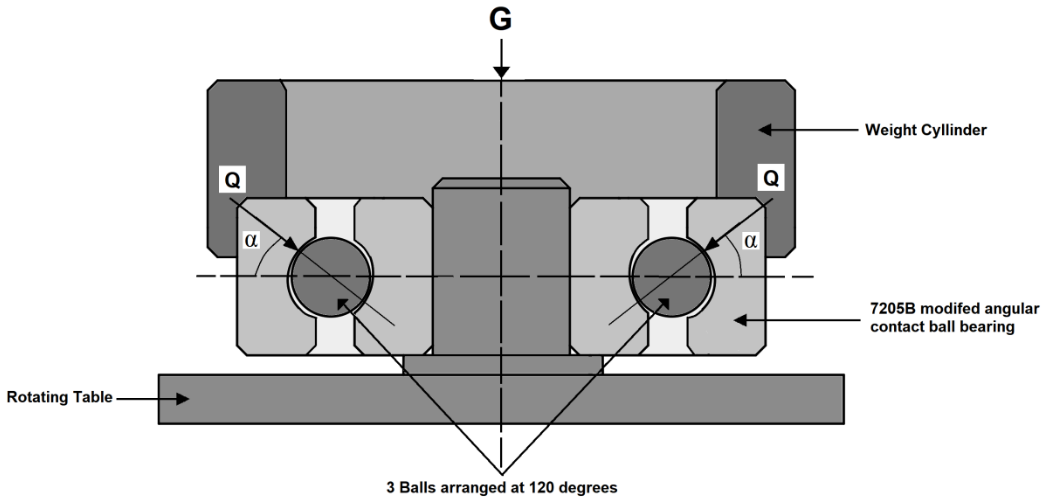

Figure 1. Modified 7205 B angular contact ball bearing with three balls and without cage.

Figure 2 presents the modified 7205 B angular contact ball bearing with three balls and with a phenol resin (textolite) cage guided on the outer race of the bearing. The clearance between cage and guided race is 0.30 mm and the clearance between balls and cage is (0.6-0.8) mm. The weight of the cage is 3.82 grams. The tests were realized for both modified ball bearings in dry conditions and with the same normal loads on ball-races contacts.

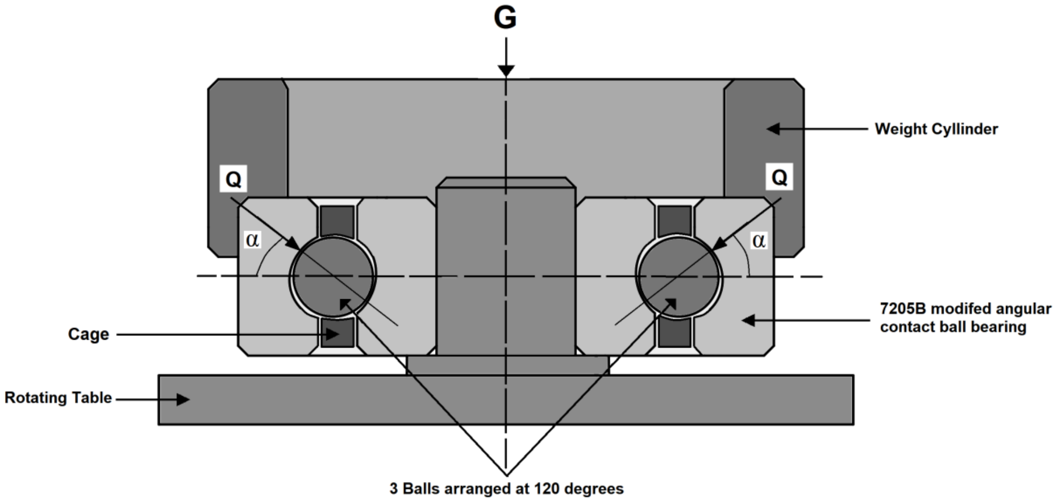

Figure 2. Modified 7205 B angular contact ball bearing with three balls and with cage.

2.2. Experimental methodology

The experiments were carried out on the Tribometer CETR UMT 2 in the Tribology Laboratory of the Mechanical Engineering Faculty of Iasi. The inner ring of the angular contact ball bearing 7205B was fixed to the rotating table of the tribometer, so that it rotates with it (Figure 1). Between the inner and outer race, three balls were placed at equidistant angles (120 degrees). The outer ring was mounted in a free rotating steel cylindrical support. The experiments were carried out with and without the cage of the bearing. Marks were placed on the steel cylinder and table in order to visualize the angular position for the two elements in rotating motion. A video camera was mounted above the assembly, capable of recording with 60 frames/second. The images captured by this camera were recorded on a computer in real time and processed with a specialized program. Details of the tribometer and 7205B modified angular contact ball bearing are presented in Figure 3 and Figure 4.

The testing methodology is similar to the spin-down methodology used for the three balls thrust ball bearings presented in [2, 3, 4]. This method consists of rotating the table up to a constant angular speed until, as a result of friction between the balls and the races, the modified bearing together with the attached cylinder will reach synchronous angular speeds (its speed will be equal to that of the table). At that moment, the rotating table of the tribometer is suddenly stopped together with the inner part of the modified bearing, while the outer part of the bearing, together with the cylinder starts the deceleration process, until it reaches a complete stop, as all the kinetic energy of the cylinder and outer part of the bearing is consumed by the friction in the six ball-races contacts. By including the cage, supplementary friction losses are developed between the balls and cage and between the cage and guide race.

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Figure 3. General view of the testing equipment.

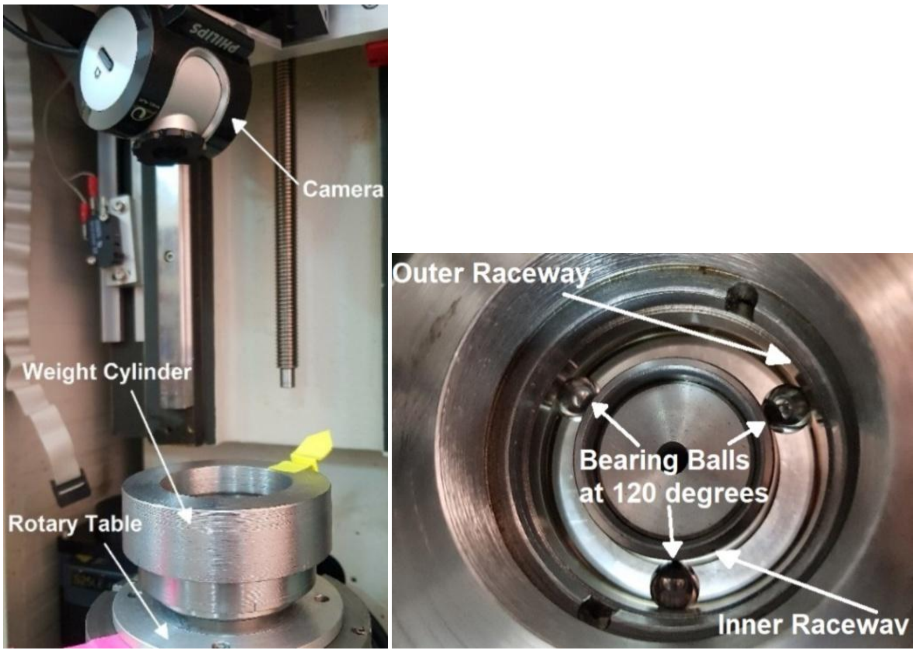

Figure 4. Details of CETR UMT 2 Tribometer and 7205B modified angular contact ball bearing.

During the deceleration process of the outer race and of the attached cylinder, the angular speed $\omega_0(t)$ decreases from an initial value ω_0 to zero, during a time t_{max}. Using a dynamic balance of the moments acting on the outer race and attached cylinder and neglecting the friction between disc and air, the following formula can be written:

$$J \cdot \frac{d\omega_o}{dt} + T_z = 0 \tag{1}$$

where J is the moment of inertia of the system defined by the outer race and attached cylinder, T_z is the total friction torque generated by the three balls in contact with the outer race.

As it was presented in papers [2, 3], in dry and limit lubrication conditions the total friction torque can be considered independent of the rotational speed and Eq. (1) can be analytically integrated having imposed the following conditions:

a. the outer race and attached cylinder stops at a time t_{max} defined by testing and the measured cumulative position angle $\varphi_{0,total}$ corresponds with the value given by the equation: $\varphi_O(t_{\max}) = \varphi_{O,total}$, where $\varphi_0(t)$ is the time variation of the angular position of the outer race measured by the overhead camera of the assembly;

b. at the initial time $t=0$, the following condition is imposed: $\omega_O(0) = \omega_O$, where $\omega_0(t)$ is the time variation of the angular speed of the outer race in the deceleration process.

3. Experimental results

The tests were realised for a rotational speed between 100 rpm and 300 rpm in dry conditions. The total friction torques acting on the outer race during the experiments were determined by the above-mentioned methodology and are presented in Figure 5 for two repeated tests with cage and three repeated tests without cage.

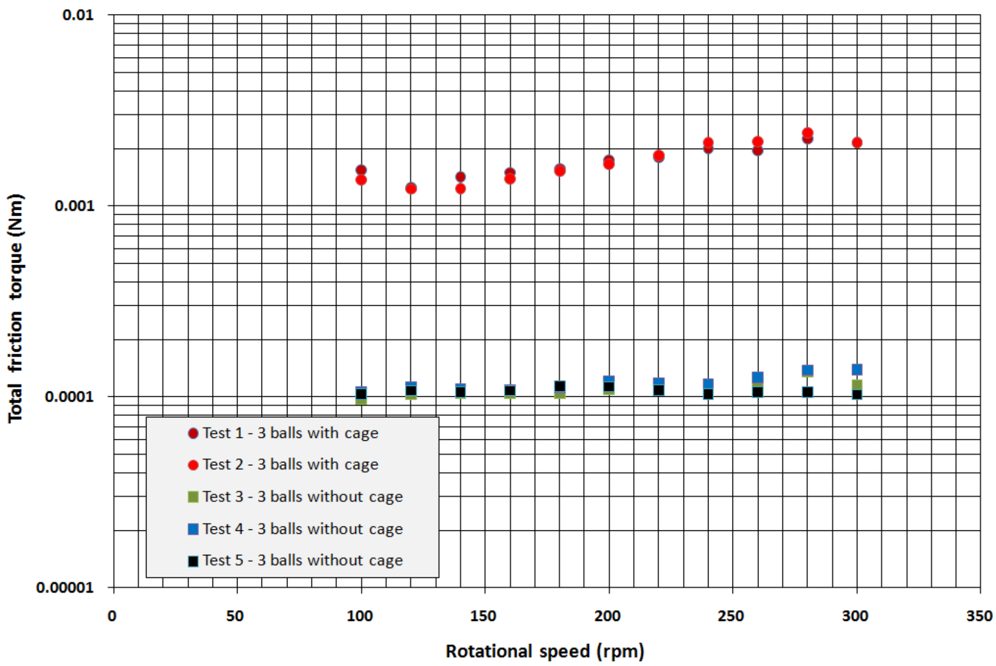

Figure 5. Experimental values of the total friction torque.

The values obtained are between 10^{-4} Nm and $1.5*10^{-4}$ Nm for the tests without the cage and between 10^{-3} Nm and $2.1*10^{-3}$ Nm for the tests with the cage.

It can be concluded that at very low loads and in absence of the lubricant, the presence of the cage leads to an increase of the angular contact ball bearing's friction torque with one order of magnitude.

4. Simulation of the friction torque in a modified 7205 ball bearing according to SKF methodology

According to SKF's proposed calculation of friction, it closely follows the actual bearing behaviour as it considers all contact areas and design changes and bearings improvements, including internal and external influences.

This model is based on the following formula [1]:

$$M = M_{rr} + M_{sl} + M_{seal} + M_{drag} \text{ [Nmm]} \tag{2}$$

where M is the total friction torque, M_{rr} is the rolling frictional moment, M_{sl} is the sliding frictional moment, M_{seal} is the frictional moment of seals and M_{drag} is the frictional moment of drag losses.

For the modified 7205 B angular contact ball bearing operating in dry conditions and without the sealing system, the total friction torque is given only by the sliding frictional component M_{sl} with following relation:

$$M = M_{sl} = G_{seal} \cdot \mu_{sl} \text{ [Nmm]} \tag{3}$$

where G_{sl} is a variable dependent on the bearing type, the bearing mean diameter dm, the axial force F_a and radial force F_r. The sliding friction coefficient μ_{sl} is determined in [1] with the following relation:

$$\mu_{sl} = \phi_{bl} \cdot \mu_{bl} + \left(1 - \mu_{bl}\right) \cdot \mu_{EHL} \tag{4}$$

where μ_{bl} is the sliding friction coefficient for limits conditions, μ_{EHL} is the sliding friction coefficient in full film conditions and ϕ_{b1} is the weighting factor for the sliding friction coefficient. For dry conditions, ϕ_{b1}=1 μ_{EHL}=0 and μ_{sl}=0.12, according to the SKF methodology [1]. To determine the parameter G_{sl} following equivalent axial force F_{ae} was used:

$$F_{ae} = 12 \cdot Q \cdot sin\left(\alpha\right) \tag{5}$$

According to SKF methodology [1], the parameter G_{sl} is determined by the equation:

$$G_{sl} = S_1 \cdot d_m^{0.26} \cdot \left[\left(F_r + F_g\right)^{4/3} + S_2 \cdot F_a^{4/3}\right] \tag{6}$$

For the given modified 7205 B angular contact ball bearing, having the normal contact load Q=6.443N, the equivalent axial force F_{ae} results F_{ae}=20.011N.

For the bearing mean diameter dm=38.5 mm and for a rotational speed n between 100 rpm and 400 rpm, it results: G_{sl}=2.479 Nmm for 100 rpm and G_{sl}=2.518 Nmm for 300 rpm.

Finaly, from Eq. (3) it results that the equivalent friction torque for a normal angular contact ball bearing 7205 B with 12 balls and cage corresponds to $2.98 \cdot 10^{-4}$ Nm and $3.02 \cdot 10^{-4}$ Nm for rotational speed of 100 rpm and 300 rpm, respectively.

Due to the fact that the tested ball bearing has only three balls, the friction torque generated by the modified bearing is:

$$M_{SKF} = \frac{M}{4} \tag{7}$$

Applying Eq. (7) to the above mentioned results of the equivalent torque M, the following values have been obtained for the theoretical SKF friction torque for the modified 7205 B angular contact bearing: $M_{SKF}=(0.74 - 0.76) \cdot 10^{-4}$ Nm.

The use of the SKF methodology in simulation of the friction torque in a modified 7205 B angular contact ball bearing having only three balls with the cage very low loaded and in absence of the lubricant, leads to theoretical values with more of one order of magnitude that the values obtained by the experiments.

5. Conclusions

The authors developed an experimental methodology to determine the friction torque in a modified 7205B angular contact ball bearing operating in dry conditions and at a very low axial load. The modified bearing has only three balls, distributed at 120 degrees between the two races. To determine only the friction torque as result of the six ball-races contacts, the experiments were performed without the cage. To evidence the influence of the cage on the friction torque, a new set of experiments have been performed with the modified 7205 bearing having three balls and a phenol resin cage guided on the outer race. As result of the experiments, the following conclusions have been formulated:

(i) For very low axial loads and dry conditions the friction torque generated only by the balls-races contacts can be considered constant between 100 and 300 rpm;

(ii) The presence of a phenol resin cage, guided on the outer race, leads to an increase of the friction torque of the modified ball bearing with more than one order of magnitude. The authors consider that, if at normal loads and speeds, the influence of the cage on the friction torque can be neglected, for very low loads, the influence of the cage on the friction torque is important and must be considered in simulation programs.

(iii) The authors simulated the friction torque for the modified angular contact ball bearing by using the SKF methodology. The friction torque resulted by the simulation program is lower with more than one order of magnitude that the measured friction torque in dry conditions. It was experimental demonstrated that, as is recommended in [1], the general methodology developed by SKF cannot be use for the simulation of the friction torque in angular contact ball bearings operating at very low loads.

6. References

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